

Design and development of an improved palm kernel shelling and sorting machine

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ABSTRACT

This paper presents a design and development of an improved palm kernel shelling and sorting machine. The varying physical and mechanical properties of the identified species of palm kernel nuts during fracture, as a means of determining the critical load required to cause fracture on the palm kernel nuts, without damage on the nut meat within were considered. The necessary evaluation of the cracking and sorting units were properly achieved through a wide range of design criteria; basically load estimation, kernel size, moisture content of shells and motion resistance of shells/kernels. These design considerations were used to calculate the required momentum necessary to achieve the needed force of cracking, and to determine the effective sorting approach. The velocity and kernel-shell characteristics evaluation are the significant factors used in the design of the optimal configurations of the shelling impeller and the sorting technique respectively. The machine optimum shelling-sorting efficiency was found to be 90 percent, throughput capacity was 59kg/h, and whole kernel recovery was 70 percent, with ease of operation, low costs of production and maintenance.

Keywords: Palm kernel, Design and development, Shelling, Sorting, Efficiency, Low costs.

1. INTRODUCTION

The oil palm tree, known as *Elaeis guineensis* Jacq., is a great economic asset. The oil palm tree is a perennial plant which is indigenous to tropical areas. It is acclaimed to be the richest vegetable oil plant [1]. The plant which originated from Africa, mostly in the southern parts of Ghana and Nigeria, but grown in plantations in Southeast Asia and Southern America, has different varieties [2], with many products

derivable from the plant, some of which are palm oil, palm kernel oil, palm kernel cake, fibre, palm wine, fatty alcohol, broom and wood plank. Within the pulp or mesocarp lies a hard-shelled nut containing the palm kernel. The palm tree grows in warm climates at altitudes within 500m above sea level, and bears its fruits in bunches which vary in weight from 10 to 40kg. The individual fruit weighing from 60 to 70gms, is made up of an outer skin (exocarp), a pulp (mesocarp) containing the palm oil in a fibrous matrix, a central nut consisting of a shell (endocarp) and the kernel which itself contains an oil, quite different from palm oil, resembling coconut oil [3]. The fruit of the oil palm is well known for its economic importance and nutritive values. Harvested palm bunches undergo processing stages of sterilisation, stripping, digestion and palm oil extraction. Palm nuts and fibres are left as residue [4]. The nuts are dried and cracked into palm kernel and shell and subsequently it is separated into palm kernel oil (PKO), palm kernel meal (PKM) and water.

With respect to the importance and merits posed by palm kernels, the demand for it in the world markets is increasing daily. Palm kernel from the cracked palm nuts are crushed in the palm kernel mill to get the palm kernel oil that is useful in making soap, glycerine, margarine, candle, pomade, oil paint, polish and medicine. The palm kernel oil is also used in the production of fuel and biodiesel. The kernel cake on the other hand serve as ingredient for livestock feeds and it is widely used in livestock industries while the fibres are used in the boiler as fuel [5-7].

Over the years, extracting and expression of oil from oil seeds involve a wide range of traditional, chemical and mechanical processes [8]. Extraction of oil from palm kernels is such an important aspect of palm kernel processing, and as the palm oil production stages in the processing line had undergone a great deal of mechanical development, the palm kernel oil production is still less mechanized and this production process actually begin with the separation of the palm nuts from the fibre. Palm oil is extracted from the pulp and the kernel oil from the kernel. Cracking palm nuts to release the kernels is therefore a critical step that affects the quality of kernel oil. Traditionally, the separation of nuts from fibre is by using a woven basket to bring out the mixture of nuts and fibre from the bottom of the processing pit, and rocking the basket back and forth to facilitate the movement of the fibre (with lower density) to the top of the nuts (with higher density) after which the fibre are packed out of the basket, thus separating the nuts from the fibre [9]. Not quite long ago, peasant farmers who abound in the trade broke the nuts, one at a time between two stones judging the magnitude of the applied force by experience. This method is slow and in addition the person cracking was in constant danger of inadvertently hitting their fingers with the stones. Preserving the kernel embedded in the palm nut when cracking the nutshell is important in the subsequent palm kernel and shell separation and, in enhancing the quality of the palm kernel oil [10]. Apart from the drudgery, time consumption and health hazards that are likewise associated with this process, addition winnowing may be necessary as sizeable quantity of fibre is still retained in the nuts. Peasant farmers break the nuts one at a time between two stones by experience.

The semi-mechanised modes of nuts cracking takes the form of hand-operated levers, as reported for Dika nuts [11]. Conventional mechanical nutcrackers are often of the centrifugal type. The nuts are either fed into a slot on a rotor turning at a very high speed or are fed into a cracking chamber where they are impacted upon by metal beaters turning at a high speed which throws the nuts against a cracking ring. The speed is adjusted for acceptable cracking efficiency. The nuts impinge the wall at random orientations but with repeated impact due to bouncing until they are discharged cracked or uncracked albeit with much kernel breakage. Palm nut in a natural rest position lies longitudinally so that the impact is applied along the lateral axis [12-13]. The knowledge of minimum impact force required for nut cracking is therefore paramount to design improvement of the existing mechanical nutcrackers [14]. The challenge of designing and actualising the successful fabrication of a motorised palm kernel sheller with lesser production time and cost, and also achieving an equivalent purpose as does the existing ones cannot just be over-emphasised. This development

is worthy of acceptance by engineers and investors as a result of the benefits derivable from the successful shelling and sorting of palm kernels, especially to countries with greater reliance on agriculture as their economy's main stay. Therefore, this work is of vital importance because it will proffer solution to the drudgery, health hazard and the inefficiency of traditional palm kernel shelling and sorting. The main rationale behind this work is to design and construct a motorised palm kernel processing (nut shelling and sorting) machine with relatively lesser production cost and time, and evaluate its performance for optimisation.

1.1 MACHINE DESCRIPTIONS AND OPERATION

The palm kernel processing machine consists of five major units: the in-feed unit, the cracking unit, the discharge outlet, the sorting unit and the driven unit.

1.1.1 The in-feed unit

The feeding unit consists of the feed hopper and the in-feed elbow. The feed hopper design was largely influenced by the throughput capacity required to make the performance of the machine satisfactory. The feed hopper is made in the shape of a frustum ($290 \times 80 \times 240\text{mm}$), and is inclined horizontally at an angle of 60° . This is to ensure freefall of the kernels through the hopper, to prevent jamming of kernels at the throat, and to make the feed hopper self-cleaning. However, the feed hopper itself is made of mild steel. The in-feed elbow is a half-parabolic tube which spans a total length of 20 inches, consisting of the hollow tube and the elbow itself. It is included in this design to prevent any form of back flow of kernels that may arise due to the sudden exposure of the kernels to the high speed of the impeller in the cracking chamber. More so, the in-feed elbow serves to present the fed kernels in such a way that each kernel is impacted by the impeller blades. The in-feed elbow is slightly tilted to an angle of 10° to the horizontal to ensure that sufficient velocity is being built up by the free falling kernels along this route before their exposure to the high velocity cracking impeller, and to further reduce the risk of jamming at the entrance of the cracking chamber. This tilting also helps in improving the efficiency of the cracking chamber, and of the machine as a whole. The in-feed elbow also is made of mild steel.

1.1.2 The cracking unit

The cracking chamber, as shown in Figure 1 (b) using computer aided design (CAD), takes the shape of a hollow cylindrical tube with rectangular (channel-shaped) impeller blades at its core. The cylinder measures $375 \times 400\text{mm}$ in its minor and major diameters respectively, and 175mm in its length. The cracking chamber is bored at a diameter of 80mm at the back surface to enable the passage of the driving shaft to the core of the chamber through the ball bearing. However, the core of the cracking chamber is characterized with the impeller tube and blades; the tube being the carriage for the rotary motion of the blades. The schematic view of the cracking chamber and its core is as shown below in Figure 1.

The cracking process is achieved by the impact force exerted on the kernels by the impeller blades against the walls of the cracking chamber. This impact force is generated by the kinetic energy of the impeller blades; the latter being facilitated by the high velocity rotary motion of the carriage tube on which the impeller blades are attached. Each kernel nut being fed to the cracking chamber is struck against the walls of the chamber by the high velocity impeller blades, thus creating sufficient impact force to loose each kernel seed from its shell covering. The cracking unit is also made of mild steel.

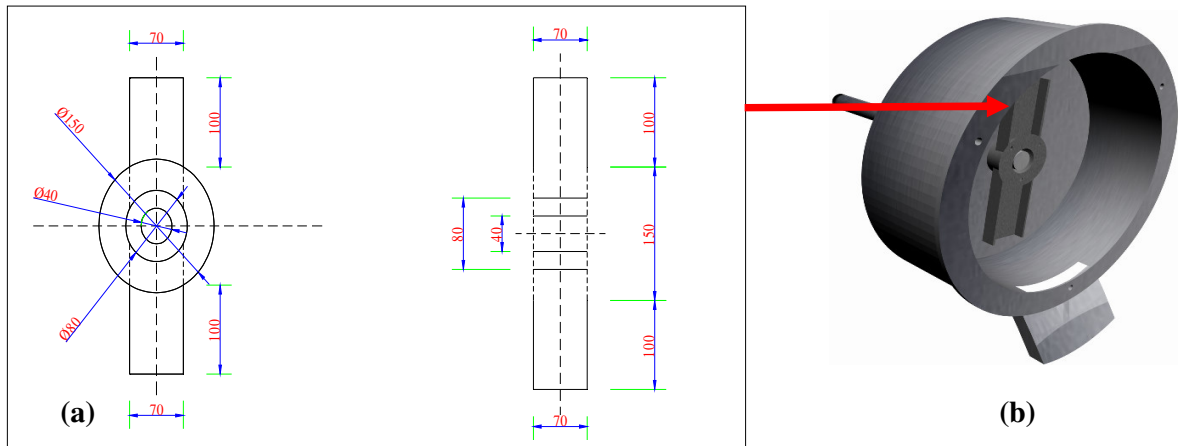


Figure 1: (a) Schematic view of the cracking impeller (b) The CAD drawing of cracking chamber

The discharge unit is situated directly below the cracking chamber. It is an opening of 180×100 mm in its width and height axes respectively. The cracked nuts are transported to the sorting unit by the passage of the discharge unit. The discharge opening was designed to allow for the passage of multiple cracked nuts per time, thus preventing jam at the discharge, and hence enhancing better sorting efficiency.

1.1.4 The sorting unit

The sorting unit is made up of a rectangular metallic mesh with uniform rectangular grooves of diameter 10 mm. This unit is directly attached to the nut outlet discharge of the cracking chamber, and it spans a total length of 400 mm, width of 180 mm, and height of 100 mm. This unit operates in the form of an agitated basket, and is stimulated by the vibration effect from the electric motor; an action which toggles it forward, backward and sideways.

The cracking chamber however, ensures that the shell coverings are effectively crushed to smaller particles compared to the kernel seeds. This crushing ensures that the variation in size between the kernel seeds and the shells is large, hence the feasibility of the sieving-separation approach. More so, the diameter of the mesh grooves selected is smaller than the average kernel seed diameter (15 mm), and this ensures that the kernel seeds are not ejected along the sorting route. Along the sorting route is a part referred to as a speed breaker. The speed breaker, A, functions to reduce the velocity of discharge of the nuts and shells from the cracking chamber, with a clearance of 20 mm from the sorting tray (Figure 2), to ensure efficient sieve-separation action along the sorting route.

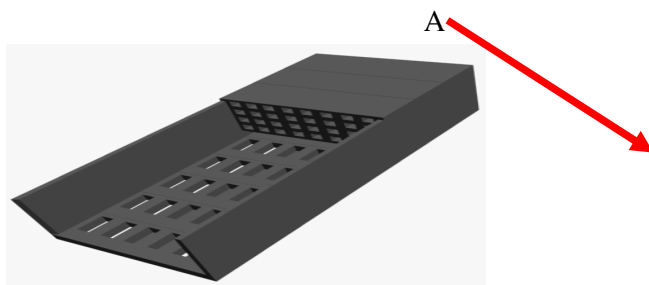


Figure 2. The CAD drawing of the sorting tray

Experimental observation proves that palm kernel seeds have a dynamic angle of repose of approximately 20° on mild steel; an angle which is lesser than the dynamic angle of repose of the shells on mild steel. This implies that the kernel seeds will develop a higher velocity coefficient along the slope, compared to their shell counterpart, and hence will avoid expulsion through the grooves. Therefore, the sorting tray is inclined at an angle of 20° to the horizontal, thus ensuring that the kernels seeds freely slide over the mesh grooves while the shells are properly expelled from the mesh grooves.

1.1.5 The driven unit

The driven unit consists of the prime mover; the electric motor, the 2 two-way pulleys and the belt drive. The electric motor is rated 3hp, with the pulleys ranging in diameter sizes of 120mm to 80mm. The belt drive is a V-belt (A60) spanning through a length of 630mm.

2. THE DESIGN ANALYSIS

2.1 The cracking unit

Kinetic energy of kernels = Impact energy of kernels on the cracking wall

$$\frac{1}{2} mv^2 = \text{Impact Energy} \quad (1)$$

But, Impact energy on the cracking wall = Work required to deform a kernel

$$\text{Work} = \frac{F}{2} \times x \quad [17] \quad (2)$$

Where F is the force or load applied, and x is the distance travelled; in this regard, the deformation on the kernels (e).

$$\text{Work required to deform a kernel (W)} = \frac{F}{2} \times e$$

$$F = P \times r \quad (3)$$

Where P is the impact loads applied to the kernels, and r is the ratio of the stress under impact to the direct stress or the deformation under impact to the corresponding deformation.

$$r = \frac{\sigma'}{\sigma} \times x \quad (4)$$

$$\sigma' = \frac{2P}{A} \text{ and } \sigma = \frac{P}{A} \quad (5)$$

Therefore, r = 2, and F = 2P

$$\text{Substituting this into } W = \frac{2P}{2} \times e$$

$$W = Pe$$

$$\text{Therefore, } \frac{1}{2} mv^2 = Pe \quad (6)$$

The product (Pe), defined as the energy of deformation, is given from experimental results as: 0.9012 and 2.0015Nm for Dura and Tenera nuts respectively [7].

2.1.1 The design of impeller

Dura Variety:

By substituting the value of mass and energy of deformation of Dura nut, the velocity required for cracking is obtained as:

$$\frac{0.00766\text{kg}}{2} v^2 = 0.9012$$

$$v = 15.33\text{m/s}$$

But $v = r\omega$; for a cracking impeller of radius, $r = 100\text{mm}$, angular velocity, ω is determined as:

$$\omega = \frac{v}{r} \quad [17] \quad (7)$$

$$\omega = \frac{15.33}{0.10} = 153.13\text{rad/s}$$

$$\text{Also, } \omega = \frac{2\pi N}{60} \quad (8)$$

$$N = \frac{60 \omega}{2\pi} = \frac{60 \times 153.3}{2\pi}$$

$$N = 1464\text{rpm}$$

Tenera Variety:

Substituting the necessary values into (6) above:

$$\frac{0.0085}{2} v^2 = 2.0015$$

$$v = 21.70\text{m/s}$$

But $v = r\omega$; and using a cracking impeller of radius 100mm .

$$\omega = \frac{v}{r} = \frac{21.70}{0.10}$$

$$\omega = 217\text{rad/s}$$

$$\omega = \frac{2\pi N}{60}$$

$$N = \frac{60 \omega}{2\pi} = \frac{60 \times 217}{2\pi}$$

$$N = 2072\text{rpm}$$

$$\text{Average linear speed required for the machine} = \frac{1}{2} (15.33 + 21.70) = 18.52\text{m/s}$$

$$\text{Average angular speed required for the machine} = \frac{1}{2} (153.3 + 217) = 185.15\text{rad/s}$$

$$\text{Average rotational speed required for the machine} = \frac{1}{2} (1464 + 2072) = 1768\text{rpm}$$

2.1.2 The shafting design

Shafts are designed on the basis of strength, rigidity and stiffness [17].

Radius of gyration (k), taking the cracking impeller and tube as a rectangular cross-section, just as shown in Figure 1:

$$k = 0.289h \quad [17] \quad (9)$$

$$k = 0.289 \times 0.35 = 0.1012\text{m}$$

But moment of inertia about the x-axis (I_{xx}):

$$I_{xx} = mk^2 \quad (10)$$

$$\text{Also } I_{xx} = \frac{bh^3}{12} \quad (11)$$

$$I_{xx} = \frac{0.07 \times 0.35^3}{12} = 2.50 \times 10^{-4}\text{m}^4$$

The mass of the cracking channel and tube referred to the axis of rotation therefore becomes:

$$m = \frac{I_{xx}}{k^2} = \frac{2.50 \times 10^{-4}}{0.1012^2}$$

$$m = 0.024 \text{ kg}$$

The tangential force (F) to the axis of rotation is given by the relation:

$$F = m\alpha \quad (12)$$

Where α is the angular acceleration; whose maximum value is given as

$$\alpha = \omega^2 r \quad (13)$$

$$F = m\omega^2 r \quad (14)$$

$$F = 0.024 \times 185.2^2 \times 0.1 = 83.76\text{N}$$

$$\text{Torque (T)} = Fr \quad (15)$$

$$T = 83.76 \times 0.10 = 8.376\text{Nm}$$

2.1.3 Mechanical power requirement

$$\text{The minimum power requirement (P)} = T\omega \quad (16)$$

$$P = 8.376 \times 185.2 = 1551.25\text{W}$$

$$P = 1.55\text{kW}$$

2.1.4 The design for strength

The shaft will be subjected to either bending stress or torsional stress or both.

$$\text{Bending stress (T}_e\text{)} = \sqrt{(M^2 + T^2)} = \sqrt{(K_m \times M_B)^2 + (K_t \times M_T)^2} \quad (17)$$

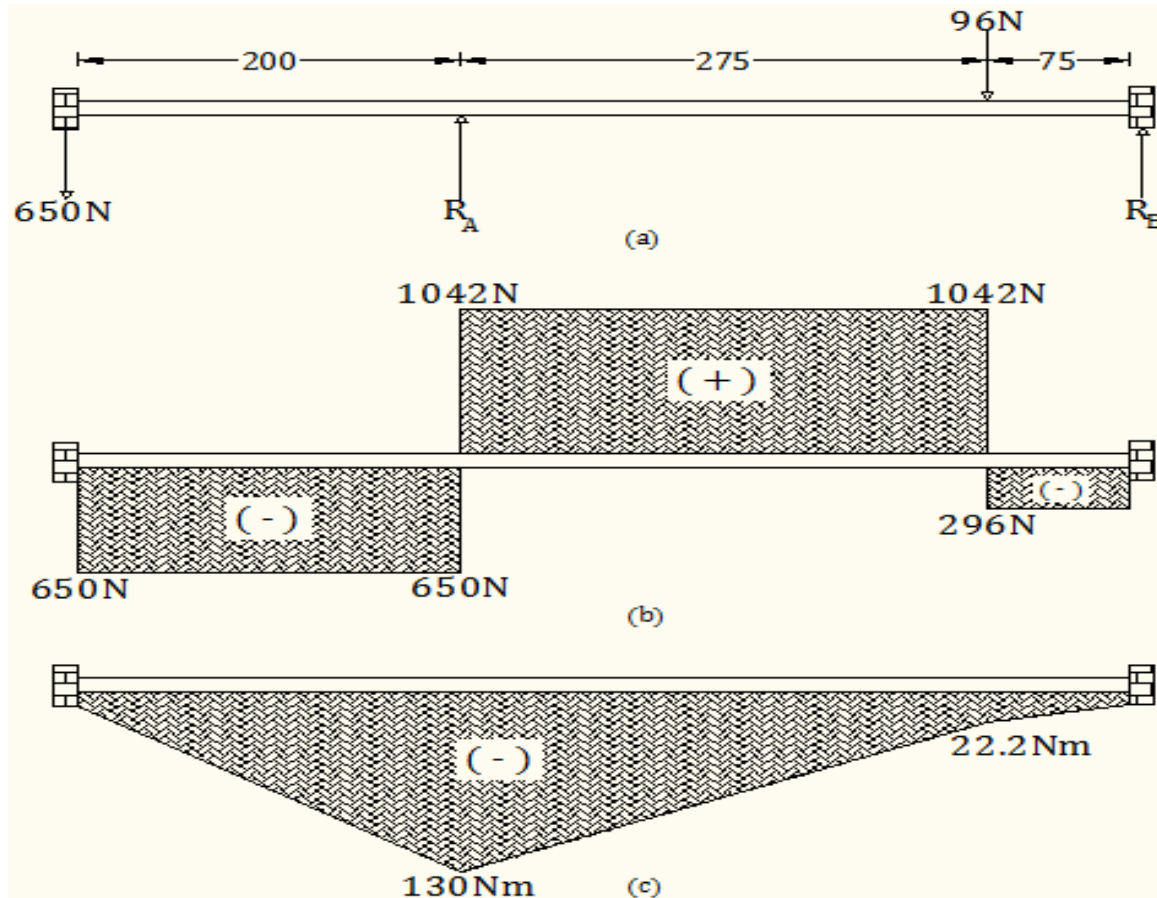


Figure 2. System of forces on the shaft, showing the shearing forces and the bending moments.

$$M \text{ (maximum bending moment)} = 130\text{Nm}$$

$$T \text{ (Torque to be transmitted by the shaft)} = 8.376\text{Nm}$$

$$M_c = \frac{1}{2} [(K_m \times M_B) + \sqrt{(K_m \times M_B)^2 + (K_t \times M_T)^2}] \quad [15] \quad (18)$$

But $K_m = 1.5$ and $K_t = 1.0$, since load consideration is assumed to be a gradually applied load (steady load)

$$M_c = \frac{1}{2} \left[(1.5 \times 130) + \sqrt{(1.5 \times 130)^2 + (1.0 \times 8.376)^2} \right] = 195.08 \text{ Nm}$$

$$\text{Also, } M_c = \frac{\pi \sigma_b d^3}{32} \quad (19)$$

$$d^3 = \frac{32 M_c}{\pi \times \sigma_b}$$

Choosing a shaft material of 0.26 carbon steel (BS 070m26) cold drawn with maximum permissible working stress, $\sigma_b = 84 \text{ MPa}$ [16]. Therefore,

$$d^3 = \frac{32 \times 195.08 \times 10^3}{\pi \times 84}$$

$$d = 28.71 \text{ mm}$$

2.1.5 The design for stiffness and rigidity (Torsional deflection)

The design for stiffness and rigidity of a shaft is determined from its torsional deflection during usage. The torsional deflection per unit length of a working shaft should not be greater than $0.25^\circ/\text{m}$. Torsional deflection of shafts:

$$\theta = \frac{TL}{GJ} \quad (20)$$

Torsional deflection per unit length:

$$\frac{\theta}{L} = \frac{T}{GJ}$$

Where T is the torsional stress on the shaft = 8.376 Nm

L = Length of shaft = 550 mm

G = Modulus of rigidity of shaft (mild steel) = 80 GNm^{-2}

J = Polar moment of inertia of shaft.

$$\text{But Polar moment of inertia of shaft, } J = \frac{\pi d^4}{32} \quad (21)$$

$$J = \frac{\pi \times 0.04^4}{32} = 2.5132 \times 10^{-7} \text{ m}^4$$

$$\frac{\theta}{L} = \frac{8.376}{80 \times 10^9 \times 2.5132 \times 10^{-7}} = 0.00042 \text{ rad/m}$$

$$\text{And } \theta = 0.00043 \times 0.55 = 0.000230 \text{ rad} = 0.00656^\circ$$

$$\frac{\theta}{L} = 0.0119^\circ/\text{m} < 0.25^\circ/\text{m}$$

The basic calculations for the sorting unit are embedded in the vibration effect required for motion along the sorting route.

2.2 The sorting unit

The following basic considerations for the sorting unit are:

Size of machine: length (L) = 1 m , width (B) = 0.46 m , height (H) = 1 m

Amplitude of vibration required on the sorting tray (δ_{ST}) = $5 \text{ mm} = 5 \times 10^{-3} \text{ m}$

2.2.1 Material selection

$$\text{Force, } F = ke, \quad (22)$$

But $F = W$ for static deflection under self-weight, and $e = \delta_{ST}$

$$W = k \delta_{ST} \quad (23)$$

$$k = \frac{W}{\delta_{ST}}$$

$$W = \rho Vg \quad (24)$$

$$W = 7830 \times (1 \times 0.46 \times 1) \times 9.81 = 35333.658N$$

But only about 2% of the volume is to be useful space in operation consideration because the structure is hollow in the width and height region, the useful weight becomes:

$$W_{\text{useful}} = 706.68N$$

$$k = \frac{706.68N}{5 \times 10^{-3}m} = 141336N/m$$

$$\text{But equivalent stiffness of the machine structure } (k_{eq}) = \frac{3EI}{L^3} \quad [19] \quad (25)$$

Since the sorting tray is a cantilever structure.

Where E = Flexural stiffness of material used to be used.

L = length of sorting tray and the machine support frame.

I = moment of inertia of the whole machine, I_{XX}

$$\text{Therefore, } E = \frac{KL^3}{3I}$$

$$I_{XX} = \frac{bh^3}{12}, \text{ considering the machine is a rectangular box with equivalent height of 500mm.}$$

$$I_{XX} = \frac{0.46 \times 0.5^3}{12} = 0.0048m^4$$

$$E = \frac{141336 \times 1^3}{3 \times 0.0048} = 0.00982GN/m$$

Therefore, a material with a minimum flexural stiffness of 0.00982GN/m was selected; the (best) material being mild steel.

2.2.2 The power for vibration

$$\text{Taking } \omega_n = \sqrt{\frac{g}{\delta_{ST}}} \quad [17] \quad (26)$$

$$\omega_n = \sqrt{\frac{9.81}{0.005}} = 44.29\text{rad/s}$$

$$r = \frac{\omega}{\omega_n} \quad [19] \quad (27)$$

$$r = \frac{185.15\text{rad/s}}{44.29\text{rad/s}} = 4.18$$

But transmissibility of amplitude:

$$\frac{X}{Y} = \frac{1}{\sqrt{(1-r^2)^2 + (2\xi r)^2}} \quad [17] \quad (28)$$

ξ = damping ratio = 2% = 0.02 for steels

$$Y = \delta_{ST} = 0.005m$$

$$r = 4.18$$

$$X = \frac{Y}{\sqrt{(1-r^2)^2 + (2\xi r)^2}}$$

$$X = \frac{0.005}{\sqrt{(1 - 4.18^2)^2 + (2 \times 0.020 \times 4.18)^2}}$$

$$X = 3.035 \times 10^{-4} \text{m}$$

$$\text{But } v = \omega X \quad (29)$$

$$v = 185.15 \times 3.035 \times 10^{-4} = 0.0562 \text{m/s}$$

$$a = \omega^2 X \quad (30)$$

$$a = 185.15^2 \times 3.035 \times 10^{-4} = 10.40 \text{m/s}^2$$

$$\text{But, Force (F) = } ma \quad (31)$$

$$\text{Mass of cracking tube and impeller(m) = } \rho V$$

$$\rho = 7830 \text{kg/m}^3$$

$$\text{Volume of hollow cracking tube and impeller}$$

$$V = \pi(D - d)^2 L + LBH \quad (32)$$

$$V = 3.142 \times (0.15 - 0.08)^2 \times 0.05 + 0.15 \times 0.07 \times 0.03 = 0.00109 \text{m}^3$$

$$m = 7830 \times 0.00109$$

$$m = 8.49 \text{kg, and } a = 10.40 \text{m/s}^2$$

$$F = 8.49 \times 10.40 = 86.63 \text{N}$$

$$\text{Torque(T) = } Fr ; \text{ where r is the radius of the cracking tube shown in Figure 1.}$$

$$T = 86.63 \times 0.04 = 3.4652 \text{Nm}$$

$$\text{Power (P) = } T\omega$$

$$P = 3.4652 \times 185.15 = 0.642 \text{kW}$$

Therefore, total power considerations for the machine:

$$= \text{Cracking power required} + \text{Power required for vibration of the whole unit}$$

$$= (1.55 + 0.642) \text{ kW} = 2.20 \text{kW}$$

A standard electric motor of 2.25kW rating was used for this design. The horsepower rating of the selected electric motor is given as 3HP electric motor with a rotational speed of 2520rpm.

Since the total power consideration for this machine has been increased from 1.55kW to 2.25kW due to the inclusion of the vibration shock needed for the sorting unit, the total torsional stress on the shaft therefore will be increased by an equal proportion, according to the linear relation between Torque (T) and Power(P) given in (16) above. Hence, the total torque on the shaft becomes:

$$T = \frac{2250}{185.18} = 12.15 \text{Nm}$$

Compensating for this torque increase of $(12.15 - 8.376 = 3.774 \text{Nm})$ on the shaft amounts to an increase in the shaft sizing from a diameter of 30mm to a diameter of 40mm.

2.3 Production cost estimation

Table 1.0: Cost of material and production

| SN | MATERIAL | UNIT | QUANTITY | UNIT PRICE ₦ (\$) | TOTAL PRICE ₦ (\$) |
|-----------|--|-------------|-----------------|--------------------------|---------------------------|
| 1 | Mild steel plate (1000 × 460 × 1000) | mm | 1pc | 11000(73) | 11000(73) |
| 2 | Hollow mild steel plate (290× 80 × 240) | mm | 1pc | 5000(30) | 5000(30) |
| 3 | Hollow mild steel drum (375 × 400 × 175) | mm | 1pc | 9000(60) | 9000(60) |
| 4 | Hollow mild steel tube (3 × 20) | in | 1pc | 3500 (23) | 3500 (23) |
| 5 | Metallic mesh (400 × 180 × 100) | mm | 1pc | 500 (3) | 500(3) |
| 6 | Bearing with housing | | 2pcs | 800 (5) | 1600 (10) |
| 7 | Shaft (40 × 550) | mm | 1pc | 1000 (11) | 1000 (11) |
| 8 | Pulley (50 × 120) | mm | 1pc | 1200 (8) | 1200 (8) |
| 9 | Pulley (50 × 80) | mm | 1pc | 1000 (6) | 1000 (6) |
| 10 | V-belt A60 (1500) | mm | 1pc | 600 (4) | 600 (4) |
| 11 | M-12 bolts and nuts | | 13pcs | 85 (0.6) | 1100 (8) |
| 12 | 3HP Electric motor (rented) | HP | 1pc | 4000 (26) | 4000 (26) |
| 13 | Labour | | | | 20000 (132) |
| 14 | Miscellaneous | | | | 6500 (44) |
| | | | | | 65500 (436) |

Table 2.0: Parts' nomenclature (list and names)

| SN | Part List | Part Name |
|-----------|------------------|------------------------------|
| 1. | A | In-feed Hopper |
| 2. | B | M12 Bolt And Nut |
| 3. | C | V-Belt |
| 4. | D | Electric Motor |
| 5. | E | Support Frame |
| 6. | F | Speed Breaker |
| 7. | G | Sorting Tray (Metallic Mesh) |
| 8. | H | Driver Pulley |
| 9. | I | Ball Bearing |
| 10. | J | Metallic Shaft |
| 11. | K | Cracking Drum |
| 12. | L | Infeed Elbow |

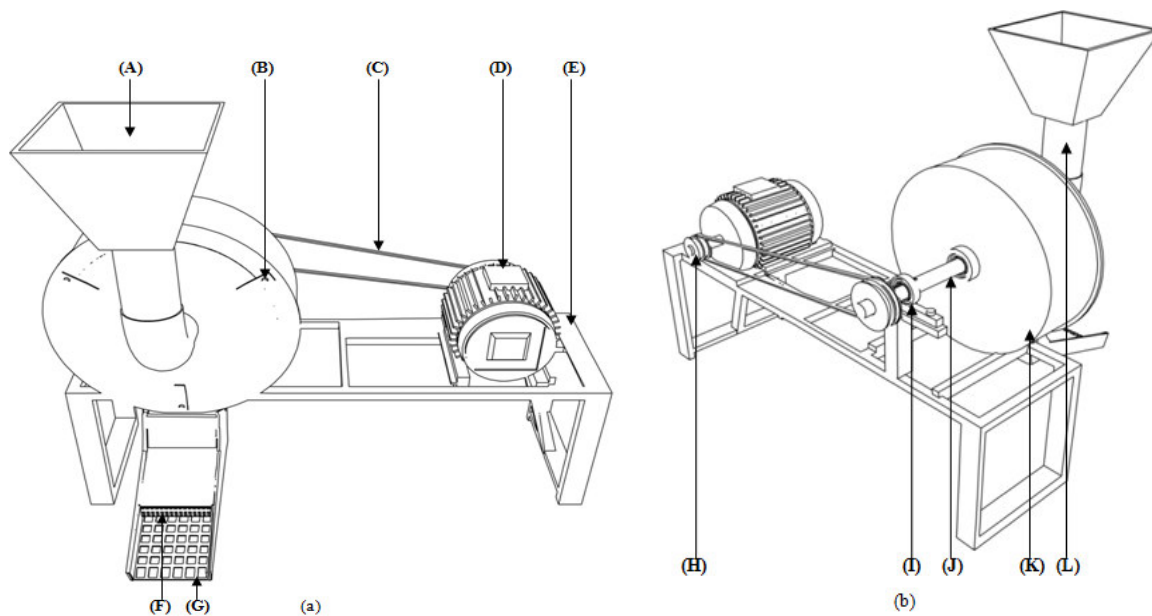


Figure 3. The CAD drawing of an isometric view of the machine.

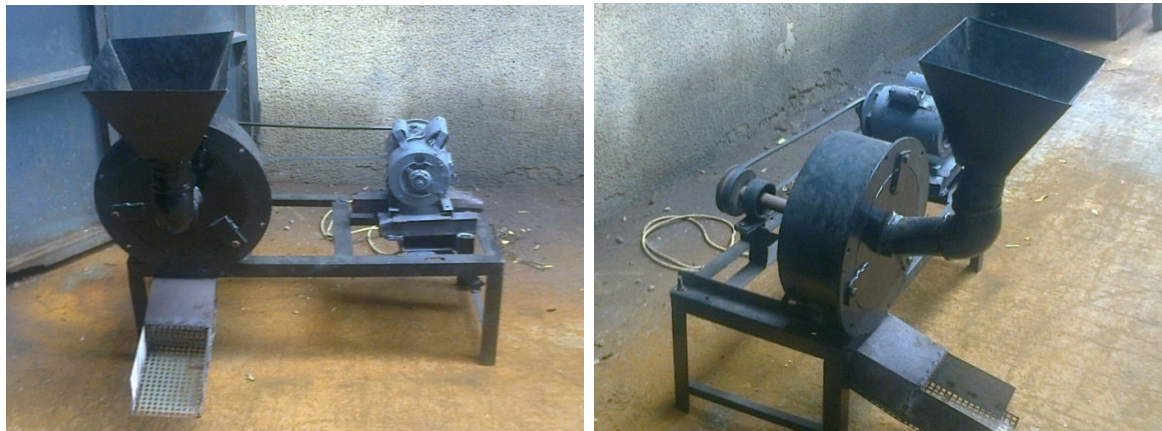


Figure 4. The improved palm kernel shelling and sorting machine

3. RESULTS AND DISCUSSION

3.1 Results

The throughput capacity of the machine was averaged over the various tests to give an average of 2.02nuts discharge per second.

Therefore, throughput capacity (nuts/hour) = Shelling rate \times 1hour [18]

= (2.02nuts/sec) \times 3600sec = 7296 nuts/hour

Throughput capacity in kg/h = Mass \times Shelling rate \times 1hour [18, 20]

Since an average palm kernel nut has a mass of 8.08g = 0.00808kg, therefore:

Throughput capacity (in kg/h) = 0.00808 \times 7296 = 59kg/h

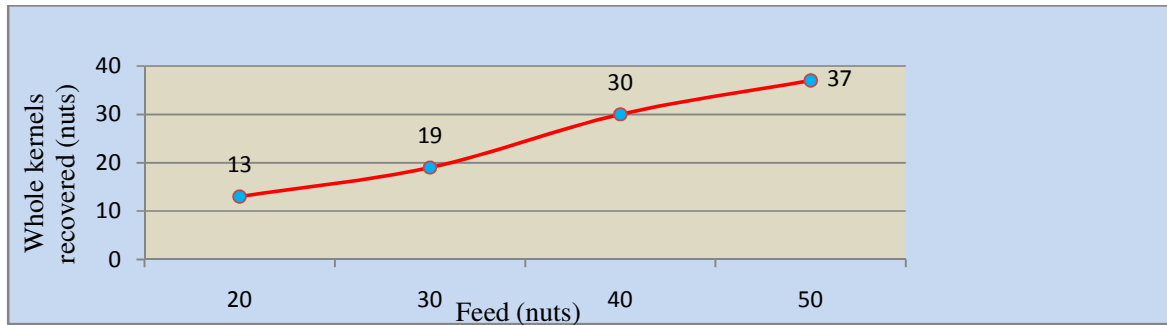


Figure 5. Whole kernels recovered against feed

$$\text{Whole kernel recovery} = \frac{\text{Whole kernels recovered (average)}}{\text{Total nut feed}} \times 100\% \quad [18]$$

$$\text{Therefore, the whole kernel recovery} = \frac{24.75}{35} \times 100\% = 70\%$$

3.1.1 Kernel size

$$\text{Efficiency} = \frac{\text{No. of completely shelled and sorted nuts (average)}}{\text{Total nut feed}} \times 100\% \quad [18]$$

$$\text{Shelling-sorting efficiency} = \frac{40}{50} = 0.80 = 80\%$$

$$\text{Effective shelling-sorting rate} = \frac{40}{19} = 2.08 \text{ nuts per second}$$

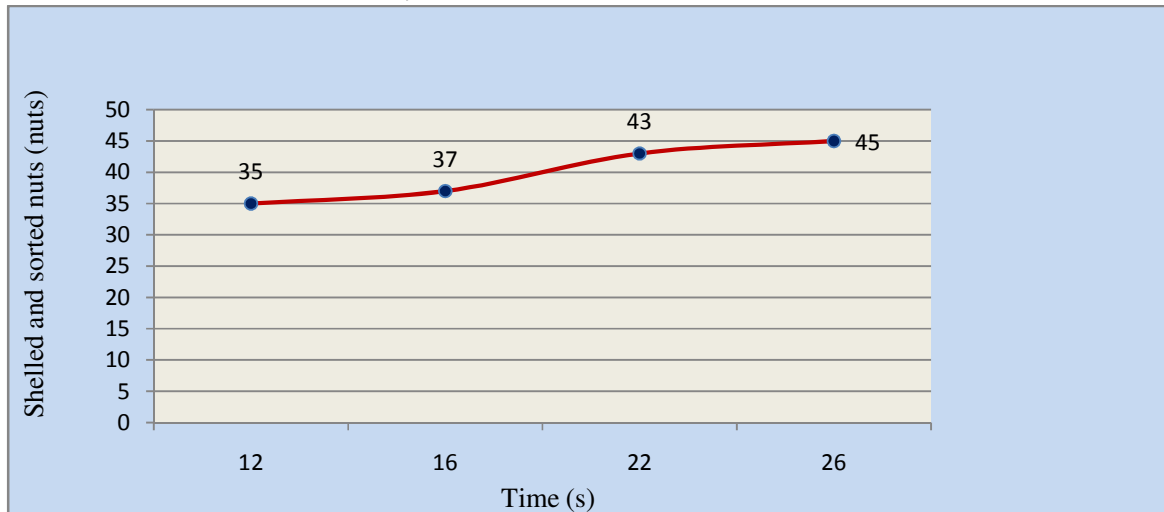


Figure 6. Shelled and sorted nuts production versus time

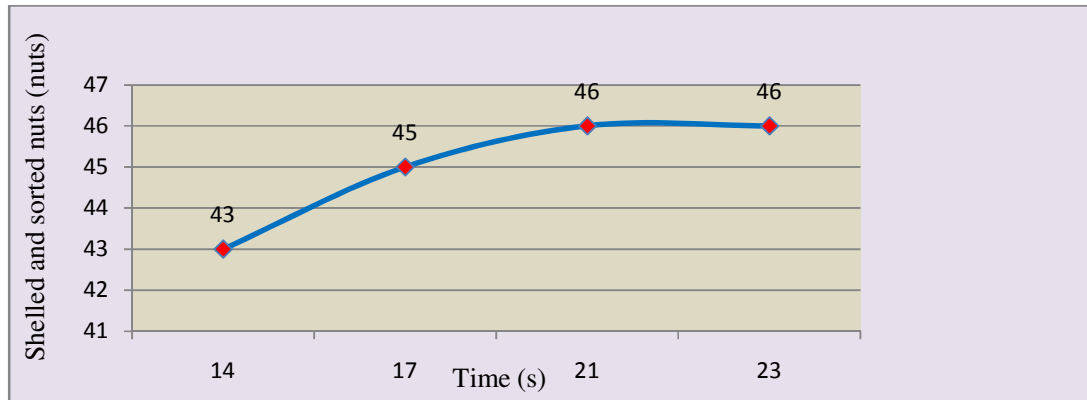
3.1.2 Moisture level of kernels

$$\text{Shelling-sorting efficiency (for the dry nut condition)} = \frac{45}{50} = 0.90 = 90\%$$

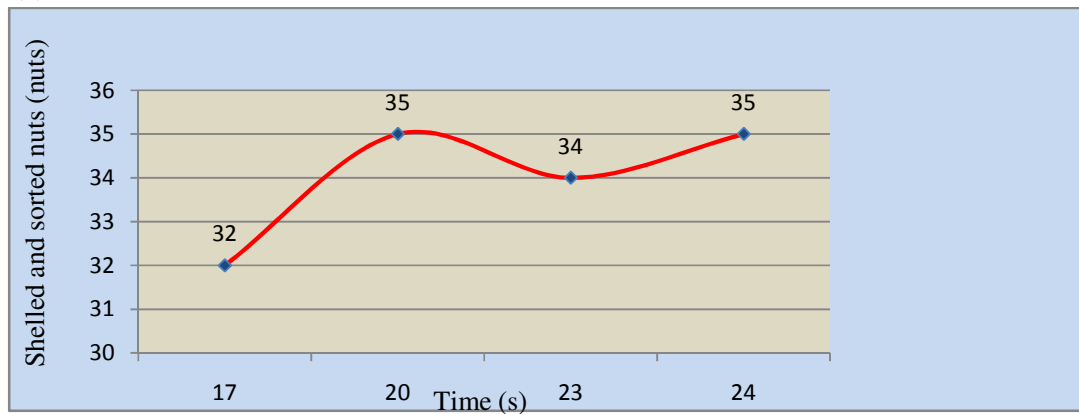
$$\text{Effective shelling-sorting rate (for the dry nut condition)} = \frac{45}{18.75} = 2.4 \text{ nuts/s}$$

$$\text{Shelling-sorting efficiency (for the wet nut condition)} = \frac{34}{50} = 68\%$$

$$\text{Effective shelling-sorting rate (for the wet nut condition)} = \frac{34}{21} = 1.6 \text{ nuts/s}$$



(a)



(b)

Figure 7 (a) and (b). Shelled- sorted nuts production versus time due to variation in moisture content. Kernel feed rate

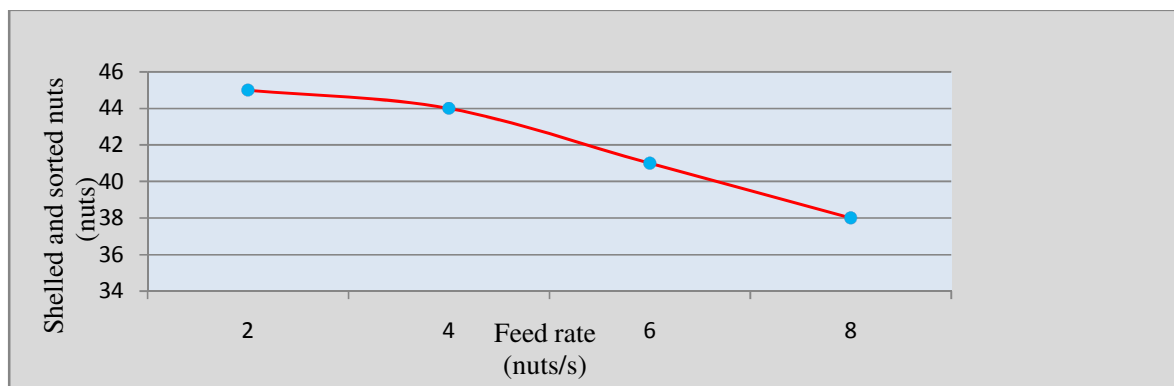


Figure 8. Shelled and sorted nuts production versus feed rate

$$\text{Shelling-sorting efficiency} = \frac{42}{50} = 0.84 = 84\%$$

3.2 Discussions

The shelling efficiency was evaluated on the basis of the number of completely shelled and sorted nuts per batch of 50 nuts for each variation in kernel size, moisture level and feed rate. The results obtained show that an average of 40 nuts were completely shelled and sorted from all variations, thus giving a shelling-sorting efficiency of 80 percent. The whole kernel recovery, which was evaluated based on the number of unbroken kernels recovered per each batch of feed, indicated an average of 24 nuts out of 35 nuts, to give a recovery

of approximately 70 percent. The throughput capacity was also obtained to be 59kg/h. The results obtained portrayed a linear dependence of the efficiency with the palm kernel size, as the machine could easily crack the palm kernels and sort the kernel seeds out accordingly with increase in kernel sizes. In another evaluation, the machine presented results that show a higher dependence of the shelling and sorting efficiency with respect to the level of moisture, as it becomes obvious that the better the dryness level of the nuts, the better the efficiency of the machine. More so, the performance of the machine shows that the rate of kernel feed into the machine; the number of palm kernels fed into the machine per unit time is a major influence on the shelling and sorting efficiency of the machine. The best of the results in this case was gotten when two nuts were fed into the machine per unit time (second), and the worst was achieved at a feed rate of eight. The results conform to the evaluation done by Eric [10] regarding the effect of moisture content of kernels.

However, the efficiency of this machine is satisfactory; giving up an efficiency range of 68 – 90%; as recorded as the performance evaluation based on the kernel sizes. Thus, the performance of the palm kernel shelling and sorting machine is largely dependent on the physical state of the kernels.

4. CONCLUSIONS

Palm kernel processing (shelling and sorting) machine of optimum shelling and sorting efficiency of 90 percent, throughput capacity of 59kg/h, and whole kernel recovery of 70 percent has been successfully and economically designed and developed. The innovation of a palm kernel shelling and sorting machine with improved qualities is a major addition to the agricultural production field of study. The machine designed and fabricated in this research was made of locally available materials, as this limited the cost of production of this machine to the barest minimum, both for peasant farmers and large scale processing industries. More so, the efficiency range and throughput capacity of the machine are satisfactory enough to ensure its adoption in the processing industries. In addition, this fabricated machine requires little or no training for its operation and maintenance.

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